Energy Conversion and Management 87 (2014) 86–95

Contents lists available at ScienceDirect

Energy Conversion and Management

journal homepage: www.elsevier.com/locate/enconman

A combined air cycle used for IC engine supercharging based on waste heat recovery

Jianqin Fu [a,b,*], Qijun Tang [a], Jingping Liu [b], Banglin Deng [b], Jing Yang [a], Renhua Feng [b]

[a] State Key Laboratory of Advanced Design and Manufacturing for Vehicle Body, Hunan University, Changsha, Hunan 410082, China
[b] Research Center for Advanced Powertrain Technology, Hunan University, Changsha, Hunan 410082, China

ARTICLE INFO

Article history:
Received 18 March 2014
Accepted 29 June 2014
Available online 21 July 2014

Keywords:
IC engine
Air cycle
Waste heat recovery
Energy saving
Supercharging

ABSTRACT

A combined air cycle is designed for internal combustion (IC) engine supercharging, which consists of IC engine working cycle and bottom cycle of waste heat recovery (WHR). The bottom cycle uses IC engine exhaust gas as cycle heat source, and its output power is used to drive the gas compressor. Both the heat transfer and thermodynamic processes of combined air cycle were investigated by numerical calculation under various cycle parameters and IC engine operating conditions. On this basis, the performances of combined air cycle and the improvement to IC engine performances were analyzed. Results show that, the cycle efficiency and exhaust gas energy recovery efficiency depend largely on the working pressure, and their maximum values appear at the working pressure of 0.35 MPa and 0.2 MPa, respectively. Compared with the naturally aspirated (NA) engine and turbocharging engine, this approach can make the fuel utilization efficiency of IC engine increase by 8.9% points and 4.1% points at most, respectively, due to the reduction of exhaust gas pressure. All these demonstrate that the proposed concept is a potentially useful approach for IC engine energy saving.

1. Introduction

At present, all countries in the world are facing increasingly severe issues of energy crisis and environmental pollution [1]. In an industrial country, more than 1/3 of the total energy consumption is caused by transportation. As a result, automobile becomes an important object for energy conservation and emissions reduction. As the main power source for automobiles, IC engine has attracted more and more attention from both scientists and engineers on its thermal efficiency [2,3]. In the study field of IC engine, the research focuses on how to improve the heat-work conversion efficiency (or total energy efficiency) of IC engine to achieve the goal of energy savings [4,5]. Referred to the literature [6–8], it can be known that a large part of fuel energy is taken away by the exhaust gas and coolant, and directly wasted in the environment. Therefore, IC engine WHR is expected to be a useful way to promote the energy utilization efficiency of IC engine, and realize the dual purposes of energy saving and emission reduction [9–11].

Up to now, a great number of scientists and scholars carried out the research on this topic for their awareness of the importance of IC engine WHR [12–15]. For example, Larsen et al. [16] have investigated the Kalina cycle for WHR on large marine diesel engines. Their research results suggest that the split-cycle process can obtain a thermal efficiency of 23.2% when using reheat compared to 20.8% for a conventional reference Kalina cycle, and reheat can increase the thermal efficiency by 3.4–5.9%. Gao et al. [17] have proposed a WHR system where a high speed turbocharged diesel engine acts as the topper of a combined cycle with exhaust gases used for a bottoming Rankine cycle. And the result shows that heat recovery system can increase the engine power output by 12%, when diesel engine operates at 80 kW/2590 rpm. Also, Boretti [18] has proposed latest concepts for combustion and WHR systems being considered for hydrogen engines, and preliminary simulations show improvement of top fuel conversion efficiencies to above 50% in the high power density operation.

Although lots of research has been conducted on IC engine WHR, the existing bottom cycles for WHR are usually independent of the IC engine working cycle. In other words, the recovered exhaust gas energy or coolant energy usually serves as other applications rather than IC engine, and there is still seldom bottom cycle used to improve the IC engine working performances. Based on this consideration, in this paper, a novel concept of combined air cycle for IC engine supercharging is proposed, which is also of interesting since it is different from the existing methods of WHR. In this combined air cycle, the bottom cycle and IC engine working cycle

* Corresponding author at: State Key Laboratory of Advanced Design and Manufacturing for Vehicle Body, Hunan University, Changsha, Hunan 410082, China. Tel.: +86 073188983108; fax: +86 073188664452.

http://dx.doi.org/10.1016/j.enconman.2014.06.097
0196-8904/

Nomenclature

P	power (kW)
$\dot{m}$	mass flow rate (kg/s)
h	specific enthalpy (kJ/kg)
η	efficiency (%)
c_P	specific heat at constant pressure (kJ/(kg K))
T	temperature (K)
p	pressure (bar)
γ	specific heat ratio
Φ	heat flux (kJ/s)
p_{me}	brake mean effective pressure (MPa)
V_s	cylinder displacement (L)
n	IC engine speed (r/min)
i	cylinder number
τ	stroke number
H_u	fuel low heating value (kJ/kg)
G_m	fuel consumption rate (kg/s)
ε	effectiveness
$q_m c$	heat capacity (W/K)
k	heat transfer coefficient (W/(m^2 K))
A	heat transfer area (m^2)

Subscripts

com	compressor
air	air working medium
bot	bottom cycle
in	intake gas
ex	exhaust gas
tur	turbine
out	output
cyc	cycle
ice	internal combustion engine
tot	total
imp	improvement
reg	regenerator
hex	heat exchanger

Abbreviation

IC	internal combustion
NA	naturally aspirated
ORC	organic Rankine cycle
BMEP	brake mean effective pressure
PMEP	pumping mean effective pressure
NTU	number of transfer units
BSFC	brake specific fuel consumption
WHR	waste heat recovery
RGF	residual gas fraction

are coupled together. The bottom cycle uses IC engine exhaust gas as cycle heat source, and its output work is used for IC engine intake gas supercharging. In this study, the recovery potential of IC engine exhaust gas energy and the improvement of IC engine performances especially thermal efficiency were discussed based on cycle calculation under various IC engine operating conditions. All these have extended the research on IC engine WHR and also provided a guidance for developing the technologies of IC engine energy saving.

2. Principles of combined air cycle

2.1. Introduction of the concept

As is well known, boosting pressure is an effectively approach to improve the IC engine power and economy performances, but the power required by gas compressor becomes a key issue. Usually, the gas compressor can be driven by various power sources, e.g., IC engine crankshaft, exhaust turbine, electric motor. According to the drive mode of gas compressor, the supercharging ways can be classified into mechanical supercharging, exhaust turbocharging, electric aided turbocharging, etc. It is wildly accepted that exhaust turbocharging is a mature technology for IC engine [19]. Actually, exhaust turbocharging is a kind of approach for IC engine exhaust gas energy recovery, since it uses exhaust gas to drive the turbine. Although the intake gas pressure is promoted through exhaust turbocharging, IC engine has to undergo a higher exhaust gas pressure due to the throttling effect in turbine [20]. In other words, the boosting pressure of intake gas is at the cost of the increase of exhaust gas pressure. As a result, IC engine should consume some effective work during the exhaust process and thus the improvement of IC engine thermal efficiency is limited. Furthermore, the increased exhaust gas pressure results in a higher residual gas fraction (RGF) in cylinder, which has negative effects on the working process of IC engine.

So as to solve these problems in traditional exhaust turbocharging, a novel approach for IC engine supercharging is proposed. As shown in Fig. 1, a set of bottom cycle system is coupled to IC engine intake and exhaust system. In reality, this bottom cycle can be regarded as a kind of regenerated Brayton air cycle [21–23]. But, it is different from the other Brayton air cycles because this bottom cycle uses IC engine exhaust gas as cycle heat resource, and the air working medium at compressor outlet is divided into two parts: part of compressed air is used for IC engine intake gas, and the remainder serves as bottom cycle working medium. Since the air working medium is used for both the bottom cycle and IC engine working cycle, and also because the two kinds of cycles are coupled together, the proposed approach is called as combined air cycle. The bottom cycle system consists of gas compressor, by-pass valve, regenerator, heat exchanger, turbine, valve, etc. Among them, the compressor (1) is utilized to compress the air working medium for both IC engine working cycle and bottom cycle; the valve (5) is used to adjust the air mass flow rate for bottom cycle; and the turbine (8) works as the energy output equipment for the recovered exhaust gas energy. When the IC engine works at start-up condition, the by-pass valve (2) is open while the valve (5) is closed. Under the circumstances, intake gas flows through the branch (alternative intake channel) rather than compressor, and the bottom cycle does not work. When the IC engine exhaust gas reaches its optimal temperature, the by-pass valve (2) is closed but the valve (5) is open. In this case, the bottom cycle begins to work. By this means, the performances of IC engine at start-up condition can be improved.

According to the theory of IC engine WHR [24], the proposed combined air cycle is an indirect method for IC engine exhaust gas energy recovery, because the IC engine exhaust gas energy is recovered through heat transfer rather than gas secondary expansion. The recovered exhaust gas energy is first converted into the effective work of turbine, and then the effective work is used to drive the gas compressor. By this means, IC engine intake gas pressure can be boosted through this combined air cycle.

2.2. Analysis of the working processes of combined air cycle

The bottom cycle processes follow the fundamental principles of regenerated Brayton air cycle, and the T-s diagram corresponding to

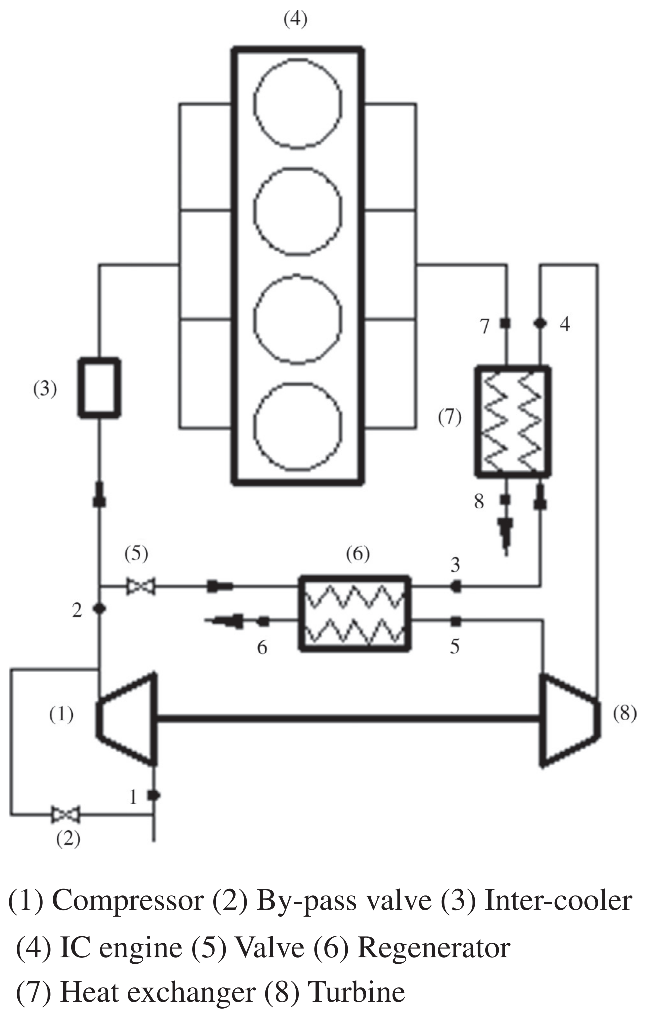


Fig. 1. Schematic diagram of combined air cycle for IC engine supercharging.

the thermodynamic processes is illustrated in Fig. 2. Combining Fig. 1 with Fig. 2, the thermodynamic processes of bottom cycle are analyzed.

Firstly, the air working medium is compressed in the compressor, and this process is expressed as 1–2(2′) in the T-s diagram. Among them, process 1–2′ represents the ideal process (isentropic compression) while process 1–2 is the real compression process.

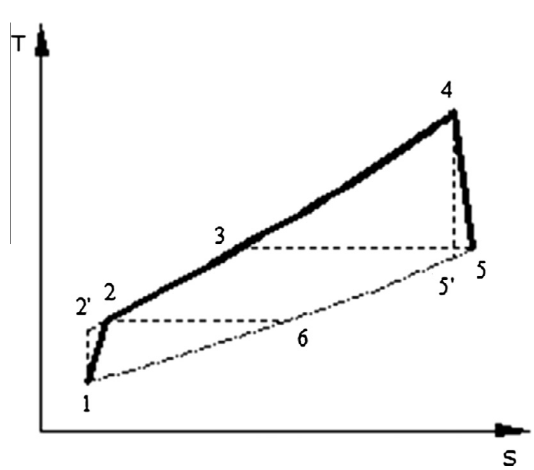


Fig. 2. T-s diagram for the bottom cycle.

The calculation formula for the compression power consumed by compressor is given as:

$$P_{com} = \dot{m}_{air} \cdot (h_{2'} - h_1)/\eta_{com} \tag{1}$$

Since the compressed air is used for both the bottom cycle and IC engine working cycle, the compressor power can also be divided into two parts according to the air mass flow rate. The compression power for the bottom cycle can be calculated as:

$$P_{com\_1} = \dot{m}_{bot} \cdot c_{P,1} \cdot T_1 \cdot \left[\left(\frac{p_2}{p_1} \right)^{\frac{\gamma_1 - 1}{\gamma_1}} - 1 \right] \Big/ \eta_{com} \tag{2}$$

And the compression power required by the IC engine intake gas can be calculated via

$$P_{com\_2} = \dot{m}_{in} \cdot c_{P,1} \cdot T_1 \cdot \left[\left(\frac{p_2}{p_1} \right)^{\frac{\gamma_1 - 1}{\gamma_1}} - 1 \right] \Big/ \eta_{com} \tag{3}$$

$$\dot{m}_{air} = \dot{m}_{bot} + \dot{m}_{in} \tag{4}$$

where P_{com} is the total compressor power, W; P_{com_1} and P_{com_2} are the compressor power consumed for the bottom cycle and IC engine intake gas, respectively, W; $\dot{m}_{air}$ is the total mass flow rate through the compressor, kg/s; $\dot{m}_{bot}$ and $\dot{m}_{in}$ are the air mass flow rate of bottom cycle and IC engine intake gas, respectively, kg/s; η_{com} is the isentropic efficiency of compressor; h_1 and $h_{2'}$ are the specific enthalpy of air working medium at state point 1 and 2′, respectively, J/kg; $c_{P,1}$ is the constant-pressure specific heat of air working medium at state point 1, J/(kg K); T_1 is the temperature of air working medium at state point 1, K; p_1 and p_2 are the pressure of air working medium at state point 1 and 2, respectively, bar; γ_1 is the specific heat ratio of air working medium at state point 1.

Then, the compressed air is pre-heated by the exhaust gas of turbine in the regenerator. As displayed in the T-s diagram, this process is assumed to be an isobaric process with the pressure loss ignored. During this process, the state of air working medium is changed from point 2 to 3. Meanwhile, the exhaust gas of turbine is cooled from state point 5 to 6. The heat exchange amount in the regenerator (from turbine exhaust gas to air working medium) can be calculated via

$$\dot{\Phi}_{reg} = \varepsilon_{reg} \cdot (q_m c)_{\min,reg} \cdot (T_5 - T_2) \tag{5}$$

where $\dot{\Phi}_{reg}$ is the heat exchange amount in the regenerator, J; $(q_m c)_{\min,reg}$ is the minimum heat capacity between the two kinds of fluids in the regenerator, W/K; T_2 and T_5 are the gas temperature at state point 2 and 5, respectively, K; ε_{reg} is the effectiveness of the regenerator; which can be calculated according to the following formulas:

$$\varepsilon = \frac{1 - \exp\left\{ (-\mathrm{NTU}) \left[1 - \frac{(q_m c)_{\min}}{(q_m c)_{\max}} \right] \right\}}{1 - \frac{(q_m c)_{\min}}{(q_m c)_{\max}} \exp\left\{ (-\mathrm{NTU}) \left[1 - \frac{(q_m c)_{\min}}{(q_m c)_{\max}} \right] \right\}} \tag{6}$$

$$\mathrm{NTU} = \frac{kA}{(q_m c)_{\min}} \tag{7}$$

where NTU is the number of transfer unit; $(q_m c)_{\min}$ is the minimum heat capacity between the two kinds of fluids and $(q_m c)_{\max}$ is the maximum heat capacity between the two kinds of fluids, W/K; k is the heat transfer coefficient, W/(m^2 K); A is the heat transfer area, m^2.

Next, the air working medium is further heated to a higher temperature by IC engine exhaust gas in the heat exchanger. This process corresponds to process 3–4 in the T-s diagram, which is also considered as an isobaric process. After this process, the state of air working medium is changed from point 3 to 4, while IC engine

exhaust gas is cooled from state point 7 to 8. During this process, the heat exchange amount in the heat exchanger (from IC engine exhaust gas to air working medium) can be calculated as:

$$\dot{\Phi}_{hex} = \varepsilon_{hex} \cdot (q_m c)_{\min,hex} \cdot (T_7 - T_3) \tag{8}$$

where $\dot{\Phi}_{hex}$ is the heat exchange amount in the heat exchanger, J; $(q_m c)_{\min,hex}$ is the minimum heat capacity between the two kinds of fluids in the heat exchanger, W/K; T_3 is the air temperature at state point 3 and T_7 is the IC engine exhaust gas temperature at state point 7, K; ε_{hex} is the effectiveness of the heat exchanger, which can also be calculated via formula (6) and (7).

After that, the compressed high-temperature air expands in the turbine. In the T-s diagram, process 4–5′ represents the ideal expansion process (isentropic process), and process 4–5 displays the real expansion process (irreversible process). During this process, the effective power is output

$$P_{tur} = \dot{m}_{bot} \cdot (h_4 - h_{5'}) \cdot \eta_{tur} \tag{9}$$

where P_{tur} is the output power of turbine, W; η_{tur} is the isentropic efficiency of turbine; $h_{5'}$ is the specific enthalpy of air working medium at state point 5′, J/kg.

In the bottom cycle, the net output power equals to the turbine output power minus the compressor power consumed for the bottom cycle, and the calculation formula is given as:

$$P_{out} = P_{tur} - P_{com\_1} \tag{10}$$

where P_{out} is the net output power of bottom cycle, W.

As two kinds of significant parameters to evaluate the cycle performances, the thermal efficiency and exhaust gas energy recovery efficiency of bottom cycle are defined in formula (11) and (12), respectively.

$$\eta_{cyc} = \frac{P_{tur} - P_{com\_1}}{\dot{\Phi}_{3-4}} = \frac{(h_4 - h_{5'}) \cdot \eta_{tur} - (h_{2'} - h_1)/\eta_{com}}{h_4 - h_3} \tag{11}$$

$$\eta_{rec} = \frac{P_{tur} - P_{com\_1}}{\dot{\Phi}_{ex}} = \frac{(h_4 - h_{5'}) \cdot \eta_{tur} - (h_{2'} - h_1)/\eta_{com}}{\dot{\Phi}_{ex}} \tag{12}$$

where η_{cyc} is the thermal efficiency of bottom cycle; η_{rec} is the exhaust gas energy recovery efficiency; $\dot{\Phi}_{ex}$ is the available exhaust gas energy, J/s, which can be calculated as:

$$\dot{\Phi}_{ex} = \dot{m}_{ex} \cdot (c_{P,7} \cdot T_7 - c_{P,0} \cdot T_0) \tag{13}$$

where $\dot{m}_{ex}$ is the mass flow rate of IC engine exhaust gas, kg/s; $c_{P,7}$ is the constant-pressure specific heat of IC engine exhaust gas at state point 7, J/(kg K); $c_{P,0}$ is the constant-pressure specific heat of IC engine exhaust gas at reference temperature, J/(kg K); T_0 is the reference temperature for calculating IC engine available exhaust gas energy.

In the meantime, the T-s diagram for IC engine ideal working cycle is shown in Fig. 3, where the compression process, combustion process and expansion process are described in process 9–10, 10–11 and 11–12, respectively. As one of the significant IC engine performance parameters, the effective power of IC engine can be calculated via

$$P_{ice} = \frac{p_{me} \cdot V_s \cdot n \cdot i}{30\tau} \tag{14}$$

where P_{ice} is the effective power of IC engine, kW; p_{me} is the brake mean effective pressure (BMEP), MPa; V_s is the displacement of each cylinder, L; n is the IC engine speed, r/min; i is the cylinder number; τ is the stroke number.

With the combined air cycle applied, the total fuel utilization efficiency of IC engine is defined as:

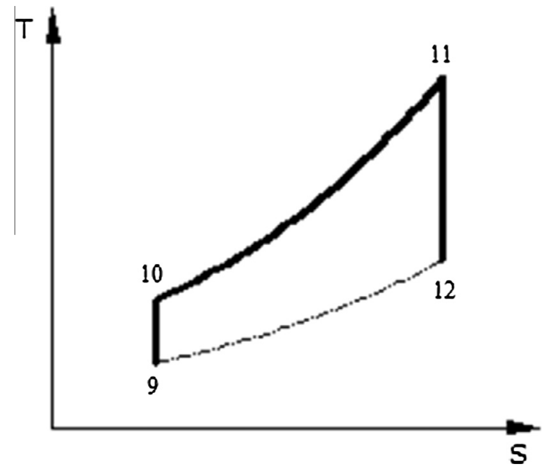


Fig. 3. T-s diagram for the IC engine working cycle.

$$\eta_{tot} = \frac{P_{ice} + P_{tur} - P_{com}}{H_u \cdot G_m} \cdot 100\% \tag{15}$$

where η_{tot} is the total fuel utilization efficiency of IC engine; H_u is the low heating value of fuel, kJ/kg; G_m is the consumption rate of fuel, kg/s.

For the purpose of evaluating the energy saving potential of this combined air cycle, the improvement to IC engine fuel utilization efficiency is defined as:

$$\eta_{imp} = \eta_{tot} - \eta_{ice} \tag{16}$$

where η_{imp} is the improvement to IC engine fuel utilization efficiency; η_{ice} is the thermal efficiency of NA engine or turbocharging engine.

2.3. Characteristics of the combined air cycle

From the viewpoint of IC engine WHR, the proposed combined air cycle is a novel approach for IC engine exhaust gas energy recovery, since the IC engine working cycle and WHR bottom cycle are coupled together, and the output power of bottom cycle is directly used for IC engine working cycle. Because the bottom cycle system uses air as working medium, it turns to be more environmental friendly and lower cost. Also because it is a kind of open cycle, the bottom cycle system seems to be much simpler, and it contributes to a lower system cost as well. From the viewpoint of IC engine supercharging, the proposed combined air cycle has lots of advantages compared with traditional exhaust turbocharging, which can be listed as follows. (1) In the exhaust turbocharging system, turbine working performances only depend on IC engine exhaust gas parameters. Since the exhaust gas parameters are changed with IC engine working conditions, exhaust turbocharger often does not work in the optimized range [20]. Unlike the exhaust turbocharging, combined air cycle can ensure the turbine usually operates in the optimal range as the working medium parameters of bottom cycle (turbine) are independent of IC engine exhaust gas. (2) In the exhaust turbocharging system, the increase of IC engine intake gas pressure results in a higher IC engine exhaust gas pressure, and it means that IC engine should consume more effective work to overcome the higher exhaust gas pressure. In this combined air cycle, the IC engine intake gas pressure (or bottom cycle working pressure) has no relation with the IC engine exhaust gas pressure. For this reason, the exhaust process work can be greatly reduced, and IC engine pumping work in the gas exchange process may change from negative to positive, which is beneficial to the IC engine thermal efficiency. (3) Furthermore,

the compressor is used for both the IC engine working cycle and the WHR bottom cycle, thus it can reduce the system cost, and make the two kinds of cycles couple together. However, just as everything has two sides, the combined air cycle also has some disadvantages, e.g., the bottom cycle working pressure range is limited by the IC engine boosting pressure.

3. Calculation for the combined air cycle

In order to reveal the energy saving potential of this combined air cycle, the proposed combined air cycle was applied to an IC engine, and the thermodynamic processes of both IC engine working cycle and WHR bottom cycle were studied. A four stroke, direct injection, water-cooled, four cylinder, exhaust turbocharged diesel engine was taken as the research object in this investigation. The detailed specifications of this diesel engine are listed in Table 1. The numerical calculation method could be developed as a useful tool to investigate the performances of IC engine coupled with combined air cycle. For the purpose of providing detailed data for numerical research, bench tests were conducted on the diesel engine and various performance parameters were obtained. In accordance with the geometry parameters and tested data of this diesel engine, numerical simulation model was established by using the simulation software GT-power, as shown in Fig. 4. That is, the IC engine working cycle was simulated by GT-power model. In this simulation model, boundary conditions of inlet and outlet were set to the parameters of standard atmospheric state. The other boundary conditions, such as mechanical friction loss, combustion efficiency, air–fuel ratio, flow coefficient of intake and exhaust valve, were calibrated by the tested data of this diesel engine.

Additionally, in order to verify the credibility of this simulation model, the model's accuracy was checked by comparing the calculated results and tested data of this diesel engine. In this study, the full load operating conditions in the entire speed range were considered. Part of the calculated results under full load, such as intake gas mass flow rate and brake specific fuel consumption (BSFC), were compared with the corresponding tested data, as depicted in Fig. 5(a) and (b), respectively. As shown, the calculation results show very good agreement with the tested data, and the relative error is so little that it demonstrates this GT-power simulation model has enough accurate to simulate the thermodynamic processes of this diesel engine. Then, the GT-power simulation model was upgraded according to the combined air cycle, and the thermodynamic processes of IC engine working cycle were simulated by the upgraded GT-power model.

At the same time, based on the calculation formulas given above, the WHR bottom cycle processes were calculated under various working parameters and IC engine operating conditions by numerical iterations. Part of the boundary conditions for WHR bottom cycle is listed in Table 2, and the remainder is provided by the calculation results of IC engine working cycle through GT-power simulation. During this calculation process, the heat transfer process in both the regenerator and heat exchanger were considered, which were calculated based on the method of efficiency-number of transfer units (NTU). The speeds of 2000 r/min and 4000 r/min were taken as two typical cases, and the calculation results under the two speeds were analyzed in the next section.

Table 1
The basic parameters of turbocharging diesel engine.

Item	Content
Engine type	Inline four cylinders, four stock
Bore (mm)	75
Stroke (mm)	89
Displacement (L)	1.573
Compression ratio	18
Ignition mode	1–3–4–2
Max torque (N m/(rpm))	230/2000
Rated power (kW/(rpm))	80/4000
Intake mode	Turbocharged
Cooling mode	Water-cooled

4. Results and discussions

4.1. Analysis of the bottom cycle performances

Before discussing the bottom cycle performances, the heat transfer processes especially the effectiveness of heat exchanger and regenerator are analyzed, since they play a very significant role in the bottom cycle performances. Fig. 6(a) and (b) shows the effectiveness of regenerator under IC engine speeds of 2000 r/min and 4000 r/min. As illustrated, the effectiveness of regenerator nearly has no relation with the bottom cycle working pressure, and it only changes with the working medium mass flow rate. More exactly, it monotonously decreases with the working medium mass flow rate. This is because the effectiveness of regenerator (heat exchanger) is mainly influenced by the heat capacity of the two kinds of fluid when the design parameters of regenerator (heat exchanger) are fixed. Because the working pressure has a very little influence on the specific heat of air working medium, it finally has a slight effect on the effectiveness of regenerator. Nevertheless, the effectiveness of heat exchanger is different from that of regenerator. As shown in Fig. 7(a) and (b), the effectiveness of heat exchanger depends on both the working medium mass flow rate and working pressure. With the increase of working medium mass flow rate, the effectiveness of heat exchanger first decreases and then increases. Because the $(q_m c)_{\min}$ in heat exchanger is changed from the air working medium to IC engine exhaust gas, it results in an inflection point in the curve of heat exchanger effectiveness. As can be seen from the two figures, the inflection points shift from low flow rate area to high flow rate area with the increase of working pressure. Unlike the effectiveness of regenerator, working pressure has great effects on the effectiveness of heat exchanger, which is also different from the research results in literature [24]. This phenomenon can be analyzed as follows. With the increase of working pressure (it is also the intake gas pressure of IC engine), both the mass flow rate and temperature of IC engine exhaust gas go up, and it leads to the change of heat capacity of hot fluid (IC engine exhaust gas) in heat exchanger as well as the effectiveness of heat exchanger. According to the analysis of heat transfer processes, it can be known that there is no practical significance to select a very high working pressure from the viewpoint of heat transfer effectiveness.

Then, the bottom cycle performances are discussed through analyzing the cycle processes. Fig. 8(a) and (b) shows the bottom cycle output power under the speeds of 2000 r/min and 4000 r/min, respectively. As illustrated, with the increase of air mass flow rate, the bottom cycle output power increases firstly and then decreases at 2000 r/min, but it changes very little at 4000 r/min. Also, the bottom cycle output power is influenced by the working pressure, and the higher working pressure leads to the larger output power. However, when the working pressure is higher than 0.3 MPa, the improvement to the bottom cycle output power is very little. In other words, there is no much significance to select a very high working pressure for the bottom cycle. In addition, since the IC engine exhaust gas energy flow is changed with the operating condition, the bottom cycle output power at 2000 r/min differs from that at 4000 r/min. Fig. 9(a) and (b) displays the thermal efficiency of this bottom cycle under the two kinds of IC engine speed. As can be seen from the two figures, the cycle

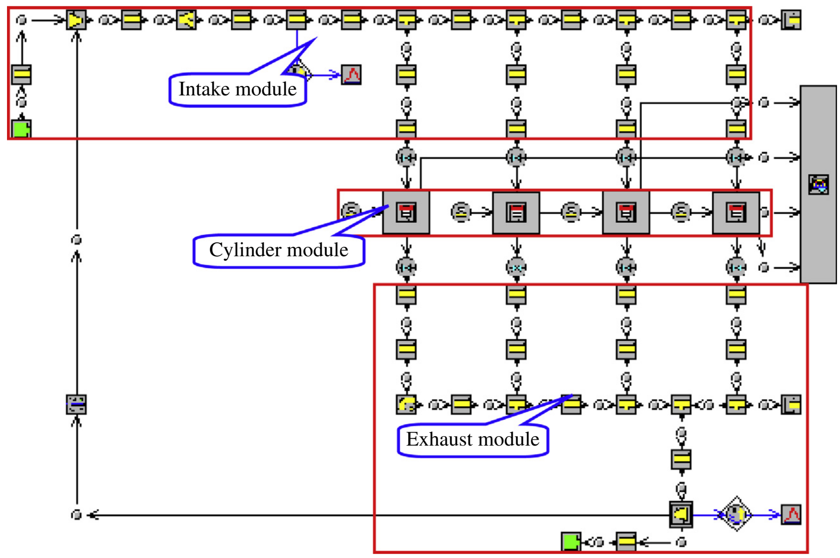


Fig. 4. Numerical model of the IC engine in GT-power.

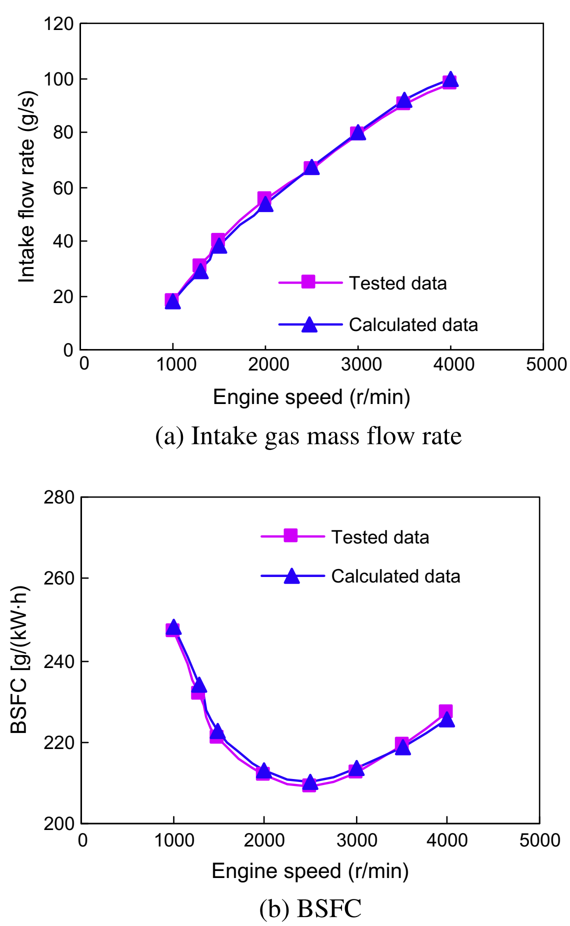


Fig. 5. Comparison of simulation results and test data.

thermal efficiency monotonically decreases with the air mass flow rate, which is different from the output power. This is because the air mass flow rate has a decisive impact on the heat transfer process between IC engine exhaust gas and air working medium. As the IC engine exhaust gas energy flow is a constant under the target operating condition, the lower mass flow rate of air working medium results in the higher working medium temperature. According to the thermodynamics theory, it can be known that the cycle efficiency relies largely on the working medium temperature. Consequently, the low mass flow rate contributes to higher working medium temperature, which is beneficial to the cycle thermal efficiency. Similar to the output power, the bottom cycle has greater thermal efficiency under higher working pressure in the investigated working pressure range. However, when the working pressure is higher than 0.3 MPa, the variation trend of cycle efficiency is very slight.

Since this combined air cycle is an approach for IC engine exhaust gas energy recovery, the recovery efficiency of exhaust gas energy is also an interesting topic. In the calculation process of exhaust gas energy flow, the reference environment temperature is assumed to be 283.15 K. Fig. 10(a) and (b) depicts the energy recovery efficiency of this bottom cycle under the two typical IC engine operating conditions. As shown in the two figures, the energy recovery efficiency ascends firstly and then descends with the air mass flow rate, which is the same as the variation trend of bottom cycle output power (see Fig. 8). Being different from the cycle thermal efficiency and output power, the energy recovery efficiency does not follow the rule of monotone increasing with the working pressure. The maximum energy recovery efficiency appears at the working pressure of 0.2 MPa or so. Moreover, the bottom cycle has higher energy recovery efficiency in the low-speed operating condition. These change rules can be analyzed as follows. On the condition that the mass flow rate of air working medium is fixed, both the compressor power and turbine power are increased as the working pressure rises. Now, the key issue is the relationship between the two kinds of power. According to Fig. 8, it can be observed that the maximum output power appears at the working pressure of 0.35 MPa. It signifies that in the working pressure range of 0.1–0.35 MPa, the increase of turbine power is higher than that of compressor power. In the meantime, with the increase of boosting pressure (which is the same as bottom cycle working pressure), both the temperature and mass flow rate of IC engine exhaust gas are also elevated, and it leads to the promotion of IC engine exhaust gas energy flow. Consequently, the increase of bottom cycle output power cannot ensure the improvement of energy recovery efficiency. As a mater of fact, when the working pressure is higher than 0.2 MPa, the improvement of exhaust gas energy flow is higher than that of output power, and it eventually causes the decrease of energy recovery efficiency.

Table 2
Boundary conditions of the combined air cycle.

Item	Content
Engine speed (r/min)	2000, 4000
Engine load	Full load
Boosting pressure (MPa)	0.15, 0.20, 0.25, 0.30
Heat exchanger: heat transfer coefficient × heat transfer area (W/K)	150
Regenerator: heat transfer coefficient × heat transfer area (W/K)	150
Turbine efficiency	0.80
Compressor efficiency	0.85
Atmospheric pressure (bar)	1.0
Atmospheric temperature (K)	283.15

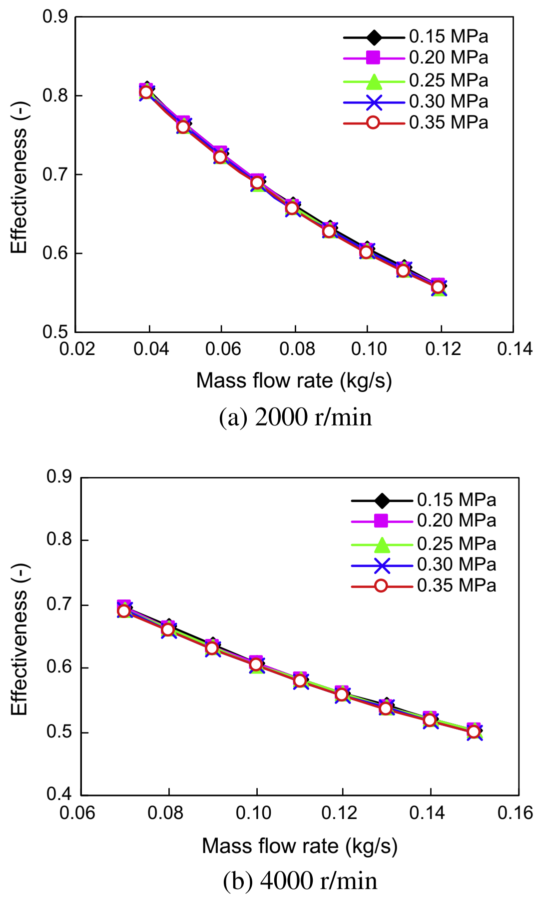


Fig. 6. The effectiveness of regenerator.

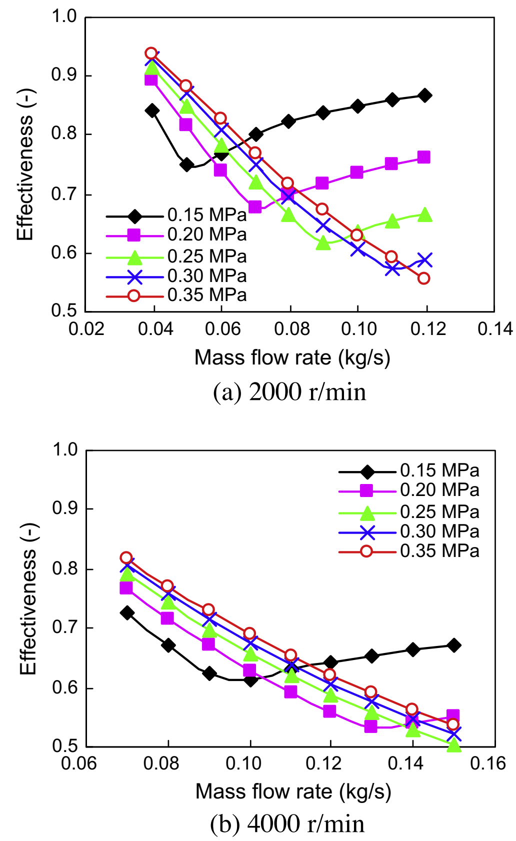
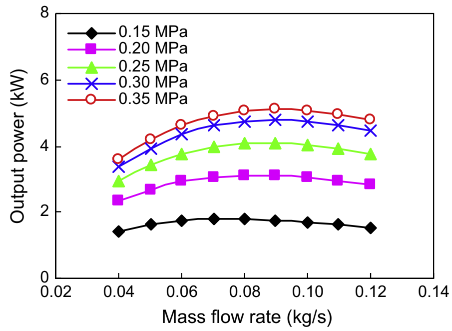


Fig. 7. The effectiveness of heat exchanger.

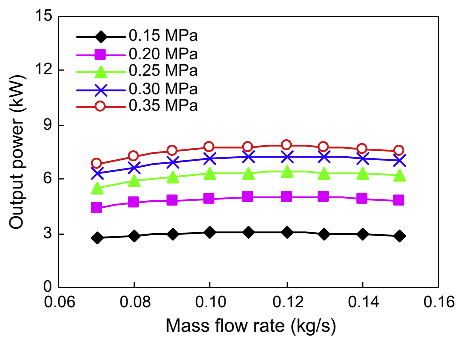
4.2. Analysis of the improvement to IC engine performances

What the most concerned about in this paper is the improvement to IC engine performances, especially the fuel utilization efficiency. According to the analysis previously, when the working pressure is higher than 0.3 MPa, the improvement to bottom cycle output power is very little. Based on this consideration, the highest working pressure (it is boosting pressure to IC engine working cycle) is set to 0.3 MPa when analyzing the improvement potential of IC engine performances. Above all, the required compressor power for IC engine intake gas is depicted in Fig. 11. As can be noticed from the figure, there is nearly a linearity relation between the required compressor power and IC engine speed. The higher the intake gas pressure is, the larger the increase rate will be. Meanwhile, the maximum bottom cycle output power under the corresponding IC engine operating condition is given in Fig. 12. Comparing Fig. 11 with Fig. 12, some conclusions can be reached. When the working pressure is at 0.15 MPa, the maximum bottom cycle output power just equals to the required compressor power in the entire speed range. Whereas, when the working pressure is higher than 0.20 MPa, the maximum bottom cycle output power is lower than the required compressor power. Under the circumstances, part of additional power (which comes from the IC engine effective work) is required to drive the compressor. In a traditional exhaust turbocharging engine, all the power required by compressor is provided by the exhaust turbine. But a large part of turbine power comes from IC engine piston work, due to the increased exhaust gas pressure [20]. In the combined air cycle, the bottom cycle output power is lower than the required compressor power for IC engine intake gas at high working pressure (larger than 0.2 MPa), however, the IC engine exhaust gas pressure as well as the exhaust process work is largely reduced compared with the exhaust turbocharging engine. Because of this reasons, IC engine pumping mean effective pressure (PMEP) is changed.

As illustrated in Fig. 13, the intake gas pressure (boosting pressure) has a crucial influence on the IC engine PMEP. In the NA engine, the PMEP is negative in the entire speed area, thus it means that IC engine has to consume a part of effective work during the gas exchange process. In the original turbocharging engine, the PMEP is also negative in most of operating conditions. In particular, the higher the IC engine speed is, the lower (negative value) the PMEP will be, owing to the increased exhaust gas pressure. But in the combined air cycle, the PMEP is changed from negative value to positive value. It means that the gas exchange processes do positive work to the IC engine. Displayed in Fig. 14 is the pumping power of IC engine under various kinds of intake gas pressure. As

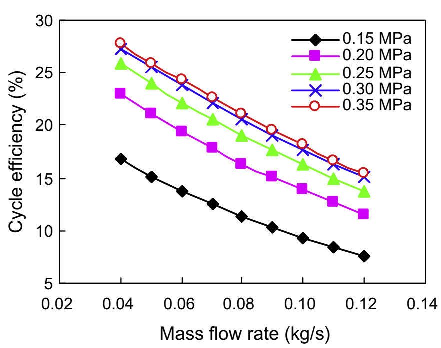


(a) 2000 r/min

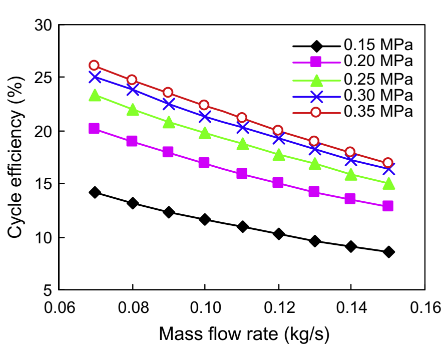


(b) 4000 r/min

Fig. 8. The output power of bottom cycle.



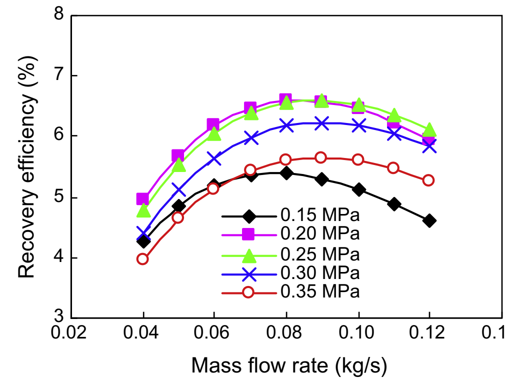
(a) 2000 r/min



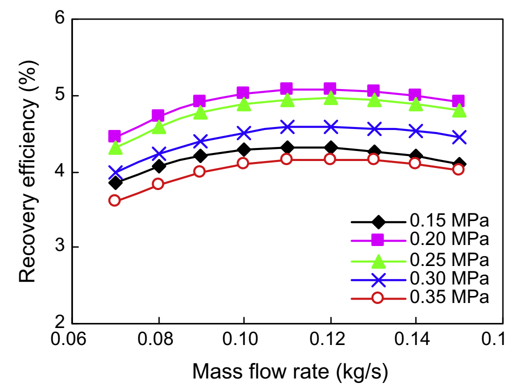
(b) 4000 r/min

Fig. 9. The cycle efficiency of bottom cycle.

it shows, when the IC engine intake gas pressure is higher than 0.2 MPa, the pumping power increases firstly and decreases later with the increase of IC engine speed. At the intake gas pressure of 0.3 MPa, the maximum pumping power comes up to 4.4 kW,



(a) 2000 r/min



(b) 4000 r/min

Fig. 10. The energy recovery efficiency of bottom cycle.

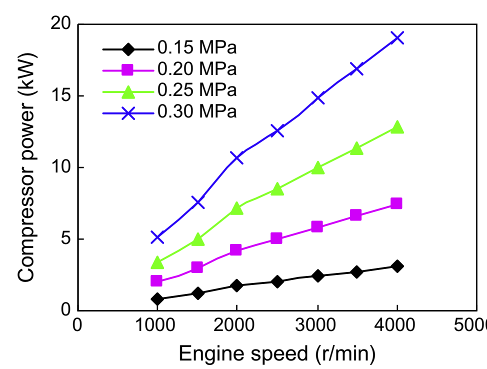


Fig. 11. The required compressor power for IC engine intake gas.

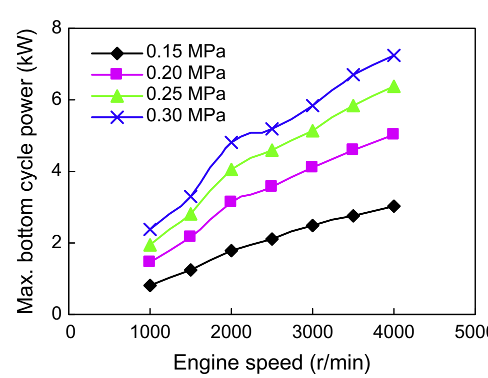


Fig. 12. The Max. bottom cycle output power.

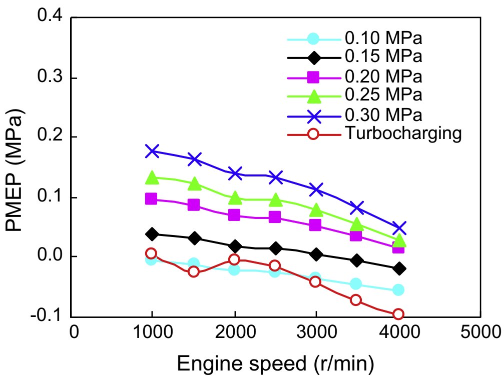


Fig. 13. The PMEP for IC engine.

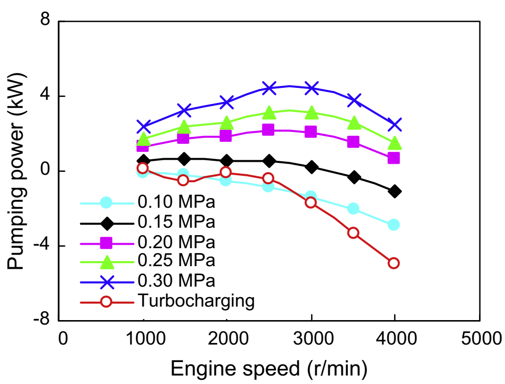


Fig. 14. The pumping power for IC engine.

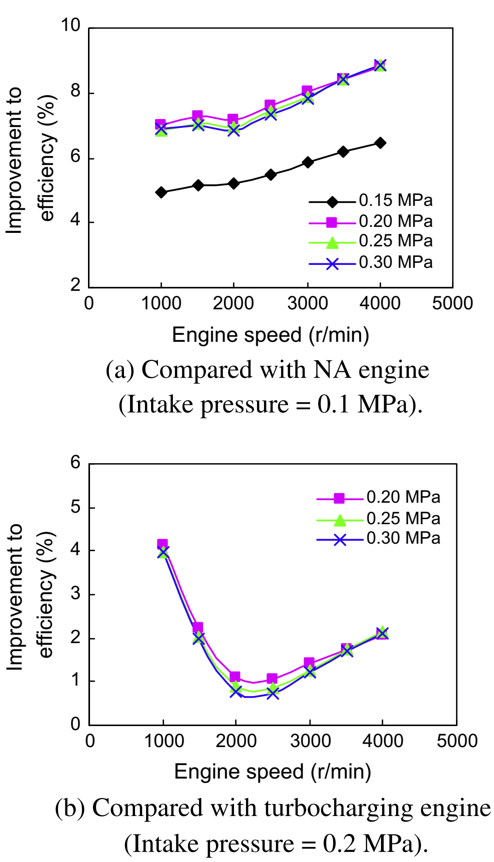


Fig. 15. The improvement to IC engine total fuel utilization efficiency.

which shows a great energy saving potential to IC engine. Eventually, the improvement to the total fuel utilization efficiency of IC engine is discussed. In this study, both the NA engine (intake gas pressure = 0.1 MPa) and the original turbocharging engine (boosting pressure = 0.2 MPa) are taken as the base engine, and the corresponding comparison results are depicted in Fig. 15(a) and (b), respectively. As shown in Fig. 15(a), with the combined air cycle applied, the total fuel utilization efficiency of IC engine has an obvious increase compared with the NA engine. The improvement to the total fuel utilization efficiency increases with the speed, and the maximum value reaches 8.9% points, while it goes up firstly and declines later as the intake gas pressure rises. Nevertheless, things are changed when compared with the original turbocharging engine (see Fig. 15(b)). With the rise of IC engine speed, the improvement to the total fuel utilization efficiency of IC engine first decreases and then increases. At 1000 r/min, it reaches the maximum value of 4.1% points. Similar to the former case, when the intake gas pressure is higher than 0.2 MPa, the improvement to the total fuel utilization efficiency of IC engine declines with the intake gas pressure. According to the above analysis of bottom cycle performances, it is not difficult to find that both the maximum recovery efficiency of IC engine exhaust gas energy and the maximum improvement to IC engine total fuel utilization efficiency appear at the working pressure (it is intake gas pressure in the IC engine working cycle) of 0.2 MPa.

5. Conclusions

In this paper, a novel concept of combined air cycle for IC engine supercharging was proposed, which is based on the principle of IC engine exhaust gas energy recovery. Compared with the traditional boosting pressure methods, e.g., exhaust turbocharging, this combined air cycle can effectively improve the fuel utilization efficiency of IC engine. Through analyzing the process of combined air cycle and comparing the performances of IC engines with/without combined air cycle, some conclusions can be drawn as follows.

Similar to exhaust turbocharging, the combined air cycle is also a boosting pressure approach based on IC engine WHR. However, the differences are in the recovery way of exhaust gas energy. In exhaust turbocharging, the exhaust gas energy is recovered by the means of gas direct expansion, while in combined air cycle, the exhaust gas energy is reused through heat transfer and thermodynamic cycle.

With the increase of working pressure (in the investigated working pressure range of 0.1–0.35 MPa), both the bottom cycle output power and cycle efficiency are improved, while the recovery efficiency of IC engine exhaust gas energy first increases and then decreases. Also, with the rise of working medium mass flow rate, both the bottom cycle output power and the recovery efficiency of IC engine exhaust gas energy increase firstly and decrease later, while the cycle efficiency monotonically decreases.

In the exhaust turbocharging system, because the increase of intake gas pressure (boosting pressure) causes the promotion of IC engine exhaust gas pressure, exhaust turbocharging engine has to consume some effective work to overcome the additional exhaust gas pressure during the gas exchange process, which has negative impacts on the IC engine PMEP and thermal efficiency. However, things are changed in the combined air cycle. With the combined air cycle adopted, the IC engine PMEP is changed from negative to positive, since the exhaust gas pressure is lower than that of exhaust turbocharging engine. As a result, the gas exchange process of combined air cycle does positive work, which is beneficial to IC engine thermal efficiency. Although the bottom cycle output power is lower than the required compressor power for IC engine intake gas at high working pressure (higher than 0.2 MPa), the increased

pumping power plays a decisive role in it. As a consequence, the total fuel utilization efficiency of IC engine can be promoted.

With the combined air cycle adopted, the energy balance of IC engine system is changed and redistributed. Owing to the change of PMEP (it is changed from negative to positive), the total fuel utilization efficiency of IC engine can be respectively promoted by 8.9% points and 4.1% points at most, when compared with the NA engine and turbocharging engine. Hence it demonstrates the proposed concept is a potentially useful approach for IC engine energy saving.

Acknowledgements

This research work is jointly sponsored by the National Basic Research Program of China (973 Program) under the Project No. 2011CB707201, the National Natural Science Foundation of China under the Project No. 51376057, and the Scholarship Award for Excellent Doctoral Student granted by Ministry of Education (No. 411198030005). The authors appreciate the anonymous reviewers and the editor for their careful reading and many constructive comments and suggestions on improving the manuscript.

References

[1] Feng RH, Yang J, Zhang DM, Deng BL, Fu JQ, Liu JP, et al. Experimental study on SI engine fuelled with butanol–gasoline blend and H_2O addition. Energy Convers Manage 2013;74:192–200.

[2] Fu JQ, Liu JP, Feng RH, Yang YP, Wang LJ, Wang Y. Energy and exergy analysis on gasoline engine based on mapping characteristics experiment. Appl Energy 2013;102:622–30.

[3] Su JY, Xu M, Li T, et al. Combined effects of cooled EGR and a higher geometric compression ratio on thermal efficiency improvement of a downsized boosted spark-ignition direct-injection engine. Energy Convers Manage 2014;78:65–73.

[4] Li T, Wu D, Xu M. Thermodynamic analysis of EGR effects on the first and second law efficiencies of a boosted spark-ignited direct-injection gasoline engine. Energy Convers Manage 2013;70:130–8.

[5] Silva C, Ross M, Farias T. Analysis and simulation of "low-cost" strategies to reduce fuel consumption and emissions in conventional gasoline light-duty vehicles. Energy Convers Manage 2009;50(2):215–22.

[6] Özcan H, Söylemez MS. Thermal balance of a LPG fuelled, four stroke SI engine with water addition. Energy Convers Manage 2006;47:570–81.

[7] Ajav EA, Singh B, Bhattacharya TK. Thermal balance of a single cylinder diesel engine operating on alternative fuels. Energy Convers Manage 2000;41:1533–41.

[8] Durgun O, Sahin Z. Theoretical investigation of heat balance in direct injection (DI) diesel engines for neat diesel fuel and gasoline fumigation. Energy Convers Manage 2009;50:43–51.

[9] Shu GQ, Liu LN, Tian H, Wei HQ, Xu X. Performance comparison and working fluid analysis of subcritical and transcritical dual-loop organic Rankine cycle (DORC) used in engine waste heat recovery. Energy Convers Manage 2013;74:35–43.

[10] Hossain SN, Bari S. Waste heat recovery from the exhaust of a diesel generator using Rankine cycle. Energy Convers Manage 2013;75:141–51.

[11] Yu C, Chau KT. Thermoelectric automotive waste heat energy recovery using maximum power point tracking. Energy Convers Manage 2009;50:1506–12.

[12] Fu JQ, Liu JP, Ren CQ, Wang LJ, Deng BL, Xu ZX. An open steam power cycle used for IC engine exhaust gas energy recovery. Energy 2012;44:544–54.

[13] Dai YP, Wang JF, Gao L. Parametric optimization and comparative study of organic Rankine cycle (ORC) for low grade waste heat recovery. Energy Convers Manage 2009;50:576–82.

[14] Fu JQ, Deng BL, Liu JP, Wang LJ, Xu ZX, Yang J, et al. Study of SI engine fueled with methanol vapor and dissociation gas based on exhaust heat dissociating methanol. Energy Convers Manage 2014;79:213–23.

[15] Wei DH, Lu XS, Lu Z, Gu JM. Performance analysis and optimization of organic Rankine cycle (ORC) for waste heat recovery. Energy Convers Manage 2007;48:1113–9.

[16] Larsen U, Nguyen TV, Knudsen T, Haglind F. System analysis and optimisation of a Kalina split-cycle for waste heat recovery on large marine diesel engines. Energy 2014;64:484–94.

[17] Gao WZ, Zhai JM, Li GH, Bian Q, Feng LM. Performance evaluation and experiment system for waste heat recovery of diesel engine. Energy 2013;55:226–35.

[18] Boretti A. Latest concepts for combustion and waste heat recovery systems being considered for hydrogen engines. Int J Hydrogen Energy 2013;38:3802–7.

[19] Kesgin U. Effect of turbocharging system on the performance of a natural gas engine. Energy Convers Manage 2005;46:11–32.

[20] Fu JQ, Liu JP, Yang YP, Ren CQ, Zhu GH. A new approach for exhaust energy recovery of internal combustion engine: steam turbocharging. Appl Therm Eng 2013;52:150–9.

[21] Göktun S, Yavuz H. Thermal efficiency of a regenerative Brayton cycle with isothermal heat addition. Energy Convers Manage 1999;40:1259–66.

[22] Zhang WL, Chen LG, Sun FR. Power and efficiency optimization for combined Brayton and inverse Brayton cycles. Appl Therm Eng 2009;29:2885–94.

[23] Wan KF, Zhang SJ, Wang J, et al. Performance of humid air turbine with exhaust gas expanded to below ambient pressure based on microturbine. Energy Convers Manage 2010;51:2127–33.

[24] Liu JP, Fu JQ, Ren CQ, Wang LJ, Xu ZX, Deng BL. Comparison and analysis of engine exhaust gas energy recovery potential through various bottom cycles. Appl Therm Eng 2013;50:1219–34.